Zhang, Y., Li, J., Sun, H., Liu, J., and Chen, Q. 2015. "Evaluation of different air distribution systems for sleeping spaces in transport vehicles," *Building and Environment*, 94: 665-675.

Evaluation of Different Air Distribution Systems for Sleeping Spaces in Transport Vehicles

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Abstract:
Air distribution in the sleeping space of a transport vehicle is important for sleep quality, especially during a long journey. In order to design a thermally comfortable sleeping space with fresher air, this investigation used a verified computational fluid dynamics (CFD) method to obtain the distributions of air velocity, air temperature, and CO_2 concentration in the sleeping space with displacement, personalized, and mixing ventilation systems. This study used the facial-area speed ratio, mean age of air, and draft risk obtained by CFD to evaluate the air distribution effectiveness. The results showed that the performance of the personalized ventilation system was better than that of the displacement and mixing ventilation systems because it provided superior thermal comfort and higher air quality. The distributions of air velocity, air temperature, and contaminant concentration computed by CFD were validated with corresponding experimental data obtained in a full-scale test rig.

Keywords: Sleeping space, CFD, Thermal comfort, Air quality, Personalized ventilation, Displacement ventilation, Mixing ventilation

Highlights:
- Thermal comfort and air quality in a small sleeping space were studied.
- The study used a validated CFD tool.
- Various ventilation systems were considered.
- Personalized ventilation is better than the mixing or displacement ventilation.

1. Introduction

Time spent sleeping accounts for one third of a person's lifetime. Sleep can help people overcome fatigue [1], enhance their immunity [2], and protect their memory [3]. People who sleep poorly may not function well in social, occupational, and educational settings [4-6]. Thus, sleep has an impact on quality of life and work efficiency. Although sleeping time on a journey may not be as long as in bedroom, good sleep in vehicles such as trains, coaches, ships, airplanes, and spacecraft is essential to overcoming fatigue and sometimes jet lag. Compared to a typical bedroom with dimensions of 4 × 3 × 3 m^3, the sleeping space in a vehicle can be quite small (2 × 1 × 1 m^3). It is very challenging to create a suitable sleeping

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environment in such a small space, because of the need to organize the air distribution so that it removes heat and CO_2 effectively without causing a draft.

Numerous investigations have been conducted on the various factors that affect quality of sleep [7, 8]. These factors include the mental and physical characteristics of a sleeping person as well as environmental factors, such as thermal conditions, air quality, acoustics, and lighting level, in the person's bedroom [9]. A number of studies have focused in particular on the effects of thermal conditions and air quality. Miyazawa [10] tracked the sleep of five high school students for 214 days and found that 23±3°C was the most suitable sleeping temperature. Furthermore, it has been found that air temperatures higher or lower than the comfortable one would decrease slow-wave sleep and rapid-eye-movement sleep, and increase the frequency and duration of wakefulness [11]. Exposure to humidity during sleep can increase wakefulness and decrease slow-wave sleep and rapid-eye-movement sleep [12]. Sekhar [13] suggested that a high level of CO_2 may shorten the duration of sleep. However, these studies all assumed a uniform sleeping environment and did not consider the impact of variations in environmental parameters on sleep quality.

Creating a suitable air distribution can make a sleeping environment more thermally comfortable and ensure good air quality. Task air conditioning and full-volume air conditioning have been widely studied in sleeping spaces because of their thermal performance and energy efficiency [9, 14-17]. However, most of the studies have focused on air distributions in large bedrooms. Very few studies are available on the environment in close proximity to a sleeping person. For example, Pan et al. [17] evaluated a bed system in which air was supplied by means of two symmetrically placed plenums on both sides of the mattress. They found that the system could save energy compared with conventional air-conditioning system used in a room. Lan et al. [14] used a personalized ventilation system that was positioned next to the head of a sleeping person. Cardiac measurements showed that personalized ventilation was better for sleep than the well-mixed ventilation for a room. Although several investigations found air distribution to be very important for thermal comfort and air quality in trains [18], aircraft [19], and spacecraft [20], very few focused on the sleeping state. The literature review suggested that it is essential to study air distributions in the sleeping spaces of transport vehicles.

Therefore, our investigation focused the thermal environment and air quality in close proximity to a sleeping space. This paper reports the results of the study.

2. Research Method

There are two primary methods of investigating air distributions in small sleeping spaces: experimental measurements [16, 17] and numerical simulations [15, 21-22], such as by the use of computational fluid dynamics (CFD). Liu et al. [23] pointed out that CFD is less expensive and more efficient for air distribution design than experimental measurements, but the modeling of turbulence can create some uncertainties. They recommended validating CFD with experimental data for the same flow characteristics before using it for design and analysis. Therefore, the present investigation has used the CFD method to evaluate the thermal environment and air quality in sleeping spaces while conducting experimental measurements of the flow characteristics for validation of the CFD results.

This investigation first used airflow data from four cases with basic flow characteristics to verify the CFD model. The verification process primarily examined the turbulence model, wall functions, and numerical algorithm that had been used in the CFD method [24]. The verified CFD method was then used to obtain the distributions of air velocity, air temperature, and CO_2 concentration in a sleeping space with different ventilation systems under various thermal and flow conditions. On the basis of the simulated distributions, this study evaluated the thermal environment and air quality in the sleeping space in terms of the facial-area speed ratio under different velocity ranges; mean age of air; and draft risk. The evaluation process identified the best air distribution for the sleeping space. This investigation then set up an experimental rig to validate the resulting design in the sleeping space. The following sections describe the research method in detail.

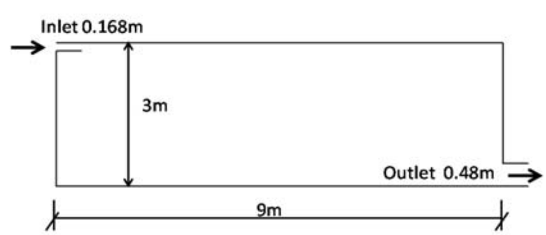
2.1 CFD method

CFD consists of direct numerical simulation, large eddy simulation, and Reynolds-averaged Navier–Stokes (RANS) modelling. Liu et al. [23] recommended using RANS modelling because it provides good results with the least amount of computing time compared with large eddy simulation and direct numerical simulation. RANS modelling has incorporated a variety of turbulence models. Zhang et al. [25] compared a number of these models and found that the RNG k–ε model [26] is stable and provides good results. Zhang's study included mixed convection with a jet supplying cool air and a warm floor that is similar to the present investigation, the flow features were the same. Therefore, the present study used the RNG k-ε model. Since this turbulence model is for high-Reynolds-number flow, our study used the standard wall function [27] for the flow near a rigid surface. The SIMPLE algorithm [28] was used to couple the pressure and velocity calculations. The standard interpolation scheme was adopted for pressure, and the second-order upwind scheme was used for all the other variables [29]. The under-relaxation factors for the pressure, momentum, energy, and concentration equations were fixed at 0.3, 0.7, 1.0, and 1.0, respectively.

This investigation simulate the flow under steady state because the flow in sleeping spaces did not change much when one was at sleep.

2.2 Verification

The CFD model described in Section 2.1 could have been used immediately in this study to calculate the air distribution in a sleeping space. However, the model incorporates many approximations to simulate airflow, and the user of the CFD program was a graduate research assistant with limited experience. According to Chen and Srebric [30], it is essential that the CFD user simulate several cases with basic flow characteristics to ensure that he/she is able to use the CFD program correctly. The airflow in a sleeping space can be considered to contain inertial force from a jet and buoyancy force from a sleeping person, which together produce mixed convection. Therefore, this investigation tested four cases with the flow characteristics shown in Figure 1: (a) two-dimensional forced convection [31], (b) two-dimensional mixed convection [32], (c) three-dimensional mixed convection [33], and (d) three-dimensional mixed convection with realistic geometry and boundary conditions [34]. These four cases represent a progressive change from simple to complex flow, and for all cases experimental data is available in the literature. The last case contains all the flow features that can be found in a sleeping space.



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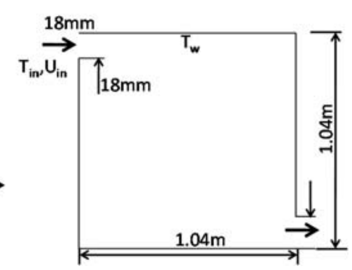

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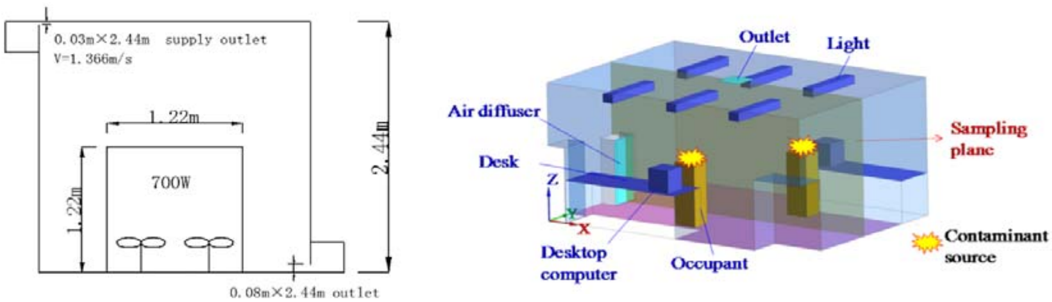


(c) (d)

Figure 1. Four cases with experimental data from the literature were used to verify the CFD program and the user's ability to use the program correctly: (a) 2D forced convection flow, (b) 2D mixed convection flow, (c) 3D mixed convection flow, and (d) 3D realistic flow.

For each of the four cases, we compared the calculated air velocity, air temperature, and CO_2 concentration profiles in the center of the room with the corresponding experimental data. Because of space limitations, Figure 2 shows the comparison only for the fourth case (Figure 1(d)). In the experiment, the CO_2 was simulated by a tracer gas. The comparison indicates reasonably good agreement between the simulated and measured results. The accuracy was similar to that reported in the literature by other experienced researchers. Although our results for the other three cases are not shown here, we again achieved similar accuracy to that reported by others in the literature. This verification process has demonstrated our ability to use the CFD program correctly to simulate airflow in a sleeping space.

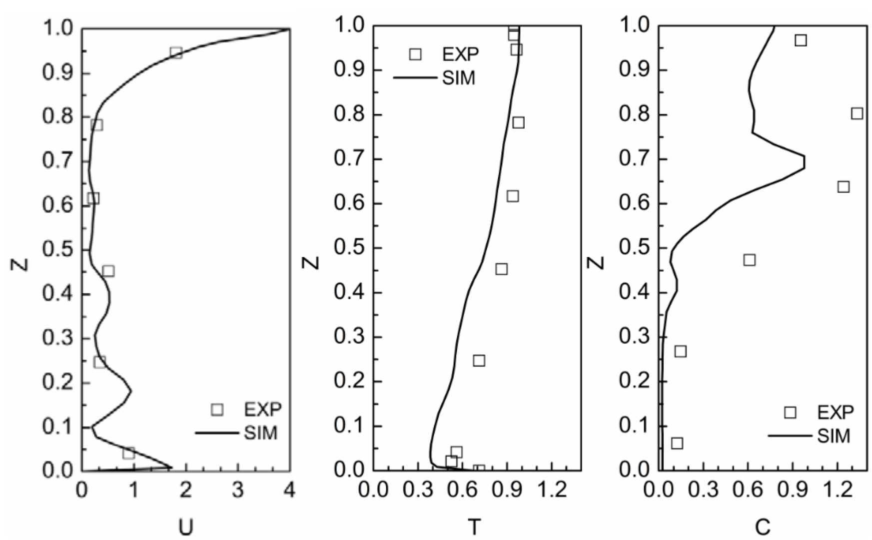


Figure 2. Comparison between simulated and measured profiles of air velocity, temperature, and tracer-gas concentration in the center of the room shown in Figure 1(d).

2.3 Evaluation of thermal environment and air quality in a sleeping space

This study used the facial-area speed ratio (FSR), mean age (MA), and draft risk (DR) to evaluate the thermal environment and air quality in a sleeping space. This section describes the use of these three evaluation criteria in the investigation.

When a sleeping person is covered with a blanket, his/her facial area becomes more sensitive. Lin and Deng [35] and Zhou et al. [36] found that the air velocity in the facial area should not be higher than 0.2 m/s to prevent draft. On the other hand, the air velocity should be high enough to prevent the accumulation of CO_2. To ensure good thermal comfort and CO_2

removal, the FSR should be as high as possible for air velocity range between 0.08 and 0.2 m/s to ensure good thermal comfort and CO_2 removal, where FSR is the proportion of the facial area with air velocity in a certain range. This study divided FSR into three air velocity ranges: ≤ 0.08 m/s, between 0.08 m/s and 0.2 m/s, and > 0.2 m/s.

If the air velocity is too high or the air temperature too low, one will feel a draft. This investigation used the following equation from ASHRAE Standard 55-2010 to calculate the DR for a sleeping environment [15]:

$$DR = (34 - t)(\mathrm{v} - 0.05)^{0.62}(0.37\,\mathrm{v}\mathrm{T_u} + 3.14) \tag{1}$$

For v ≤ 0.05 m/s, v = 0.05 m/s, and for DR > 100%, DR = 100%. In this equation, t is air temperature, v air velocity, and T_u turbulence intensity.

To evaluate the air quality in a sleeping space, this investigation used the MA, which is the time needed for the air from a supply outlet to reach the point of interest. The younger the MA is, the fresher the air would be. MA can be calculated by [37]:

$$U_\mathrm{i}\frac{\partial\bar{\tau}}{\partial \mathrm{x_i}} = \frac{\partial}{\partial \mathrm{x_i}}[(\frac{\nu_t}{\sigma_{\bar{\tau}}} + \frac{\nu}{\sigma})\frac{\partial\bar{\tau}}{\partial \mathrm{x_i}}] + 1 \tag{2}$$

where $\bar{\tau}$ is MA, ν laminar viscosity, ν_t turbulent viscosity, σ the laminar Schmidt number, and σ_t the turbulent Schmidt number. This study calculated the diffusion coefficient, Γ, for MA by using the following equation [38]:

$$\Gamma_{\bar{\tau}} = 2.88 \times 10^{-5}\rho + \frac{\mu_\mathrm{eff}}{0.7} \tag{3}$$

where μ_eff is the effective viscosity of the air and ρ the air density.

2.4 Experimental measurements

After using CFD to identify the best ventilation system for a sleeping space, this investigation built a full-scale test rig of the sleeping space, as shown in Figure 3(a). To facilitate the PIV measurements, the model was constructed from transparent acrylic resin to allow for visual access. A thermal manikin with a blanket was heated by resistance wires with a power input of 52 W, according to the recommendation by Ning et al. [39] for simulation of a sleeping person. To simulate the CO_2 exhaled by the person, this study used sulfur hexafluoride (SF_6) as a tracer gas. Since the background SF_6 concentration is zero, use of the tracer gas eliminated the effect that background CO_2 would have had on measurements of exhaled CO_2. As shown in Figure 3(b), SF_6 was released through a porous plastic ball on the mouth of the manikin at a constant rate of 6×10^{-4} m^3/h to simulate constant breathing. The SF_6 supply rate led to a minimum concentration of 0.006 ppm that can be detected by our instrument. The SF_6 concentration can be easily scaled up to become CO_2 concentration if necessary. However, for validation the CFD results, this investigation used directly SF_6 concentration for comparison.

Since the flow rate of SF_6 was very low through the porous ball, the momentum effect from the breathing was negligible. As the tracer gas contained only 1% SF_6 and 99% N_2, the impact of the density difference on the airflow was negligible. The test rig was placed in a laboratory where the air temperature was maintained with less than 1 K fluctuation.

The air was supplied to the sleeping space by a fan that controlled the flow rate of the supply air. Thus, the temperature outside the sleeping space and the temperature of supply air were the same. Due to the large flow rate and small temperature difference between inside and outside of the sleeping space, the heat transfer through the sleeping box enclosure was negligible. However we did measure the surface temperature for the walls as well as manikin surfaces, the temperatures were used as boundary conditions for the CFD simulation so that a radiation model was not needed.

(a) (b)

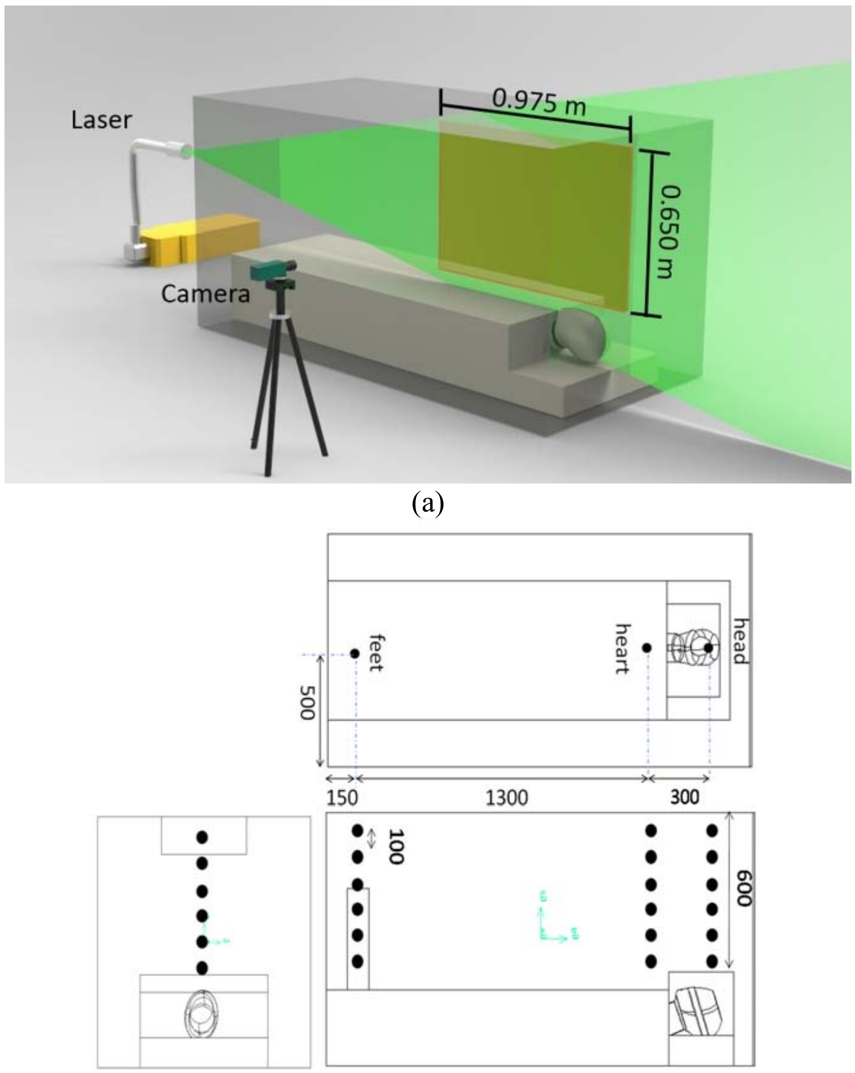
Figure 3. (a) Full-scale test rig for the sleeping space and (b) SF_6 source on the mouth of the sleeping manikin.

The supply air velocity was measured by an anemometer (Model TSI 8386) with an accuracy of ±3.0%. The surface temperature of the thermal manikin was measured by an infrared camera with an accuracy of ±1.5 K. The temperatures of the wall surfaces and supply air were measured by T-type thermocouples with an accuracy of ±0.5K. The SF_6 concentrations were measured by a photoacoustic gas monitor (Model INNOVA1412) with an accuracy of ±2%.A particle imaging velocimetry (PIV) system was utilized to measure the air velocity distribution in the sleeping space. Figure 4 shows the PIV system and the section used for measurements. Table 1 lists the key parameters of the PIV system. An adaptive correlation algorithm with a highly accurate sub-pixel interpolation scheme, which is included in the Dantec software package DynamicStudio v3.41, was used to extract the velocity vectors from the aquired particle image pairs. Then, to obtain the time-averaged airflow field, 360 uncorrelated instantaneous velocity fields were measured in each conditions at a sampling frequency of 3 Hz. The air temperature and SF_6 concentration in the sleeping space were measured by the T-type thermocouples and photoacoustic gas analyzer, respectively. The measurements were conducted at the head, heart and feet positions at six different heights in the mid-section of the sleeping space, as shown in Figure 4(b).We monitored the temperatures of supply air and feet, when the temperatures didn't change with time, we considered it was steady and the waiting time was about 3 hour.

Table 1. Key parameters of the PIV system

Item	Key parameter
Laser source	YAG double pulse laser
CCD model	FlowSense EO (11M pixels 3fps)
Lens	Nikon Nikkor 35 mm lens
Laser power	350 mJ/pulse
Time between pulses	2000 ms
Visual field of CCD	4032 pixel × 2688 pixel
Size of interrogation window	64 pixel × 64 pixel
Overlap	25%

Dimension of view field	0.950 m × 0.650 m
Sampling frequency	3 Hz
Tracing particles	DEHS
Diameter of tracing particles	1 μm



(a)

(b)

Figure 4. (a) PIV system setup and (b) temperature and SF_6 concentration measurement locations.

3. Results

3.1 Case setup

Figure 5 shows the sleeping space used in this study, which was 2.0 m long, 1.0 m wide, and 1 m high. The investigation considered three different ventilation systems: personalized ventilation (PV), displacement ventilation (DV), and mixing ventilation (MV). Figure 5 also shows the air supply locations and sizes of these ventilation systems, and the location and size of the outlet used by all three systems. Only one of the systems was operated a given time so that different air distributions could be realized. The breathing mouth was simplified to a round opening with an area of 1.4 cm^2. Table 2 summarizes the boundary conditions used for the CFD simulations. In the simulated case of validation, the wall temperatures were measured by thermocouples and specified in CFD modeling. However, in the 21 cases that were designed for the comparative study, the wall of sleeping space was assumed to be adiabatic wall.

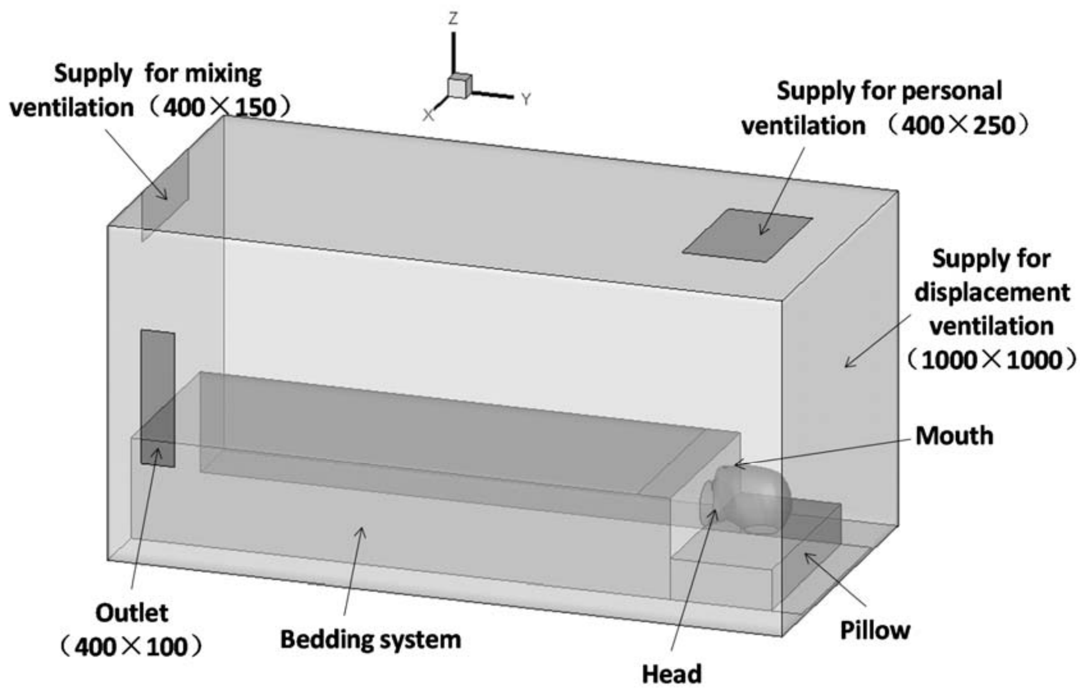


Figure 5. Schematic of the sleeping space

Table 2. Boundary conditions used in the CFD simulations

Boundary element	Conditions
Supply outlet	Airflow rates from 1.5 to 3 m^3/min Air temperatures from 20 to 26^{o}C CO_2 concentration at 500 ppm Turbulence intensity at 5%
Mouth of the sleeping person	Steady airflow rate at 6 l/min Exhaled air temperature at 34^{o}C CO_2 concentration at 36,000 ppm Turbulence intensity at 5% Hydraulic diameter at 0.014 m [40]
Head	Rigid surface at 34.6 ^{o}C
Pillow	Rigid, adiabatic surface
Body of the sleeping person	Rigid surface at 25 ^{o}C
Outlet	Outflow
Walls	Rigid, adiabatic surface

3.2 Grid mesh for CFD simulations

CFD simulations require a suitable mesh type and size to reduce computing costs while obtaining accurate numerical results. Because a high mesh quality was required for such a complicated geometry as the sleeping space [41], our CFD simulations used a tetrahedral mesh. Figure 6 shows the details of the meshes with 1.62 million cells that were used for the sleeping space.

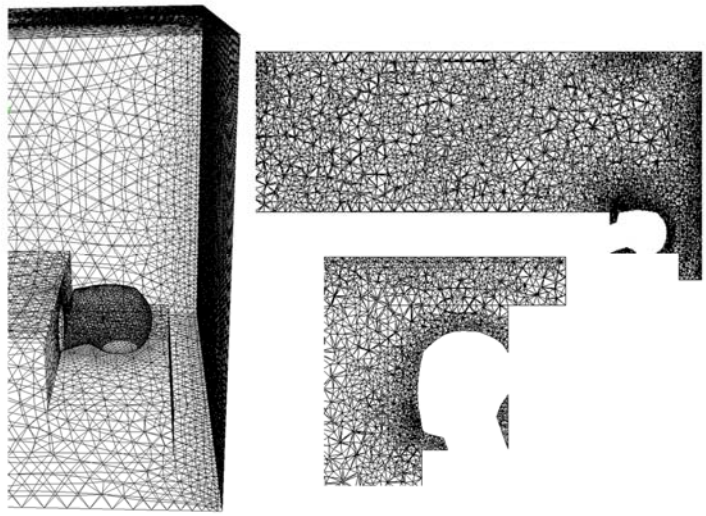


Figure 6. Mesh distribution with 1.62 million cells used for the sleeping space.

To determine the minimum mesh size required, this investigation conducted a grid independence study with meshes of 0.42, 1.62, and 6.54 million cells. In grid independence tests, one typically would double the grid number in one direction for Cartesian coordinate. This would increase the grid number by 2^3 (=8) times. Since this study used unstructured grid, it has also three directions (three dimensional), the grid number should be increased by roughly 8 times. Since our baseline grid number was 0.42 million, the other two should be 3.36 million and 26.88 million. A grid of 26.88 million exceeded our computer capacity. That is why 0.42, 1.62 and 6.54 million were used.

We compared the vertical air velocity profiles calculated with these three grid numbers at the head, heart, and feet positions of the sleeping person (Figure 7). The comparison suggested that 1.62 million cells would be sufficient for this case. With 1.62 million cells, the mesh size was 5 mm for the face and mouth area, 10 mm for the supply outlet, and a maximum of 50 mm in the other areas of the sleeping space.

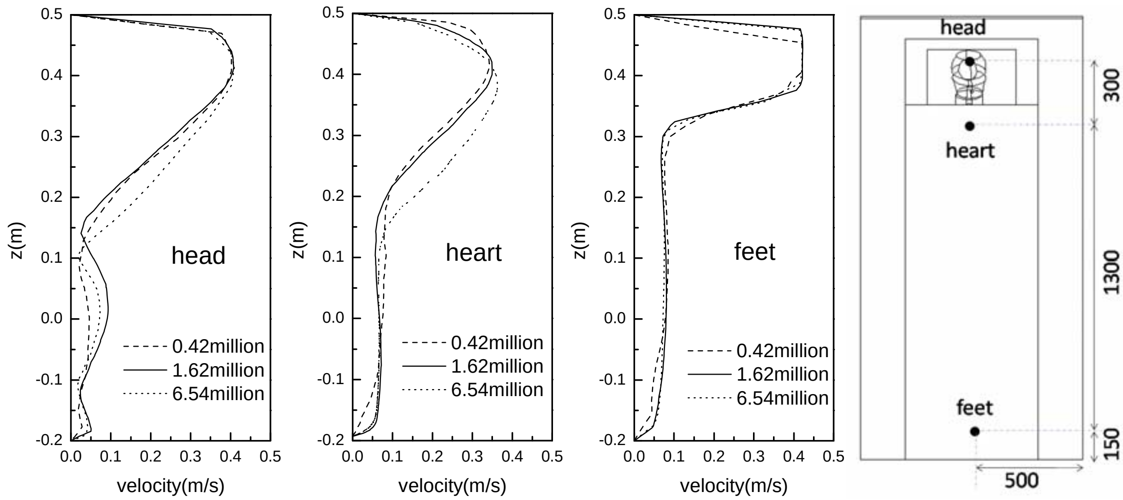


Figure 7. Comparison of the vertical air velocity profiles calculated with different grid numbers at the head, heart, and feet positions of the sleeping person.

3.3 Evaluation of the ventilation systems and air supply parameters

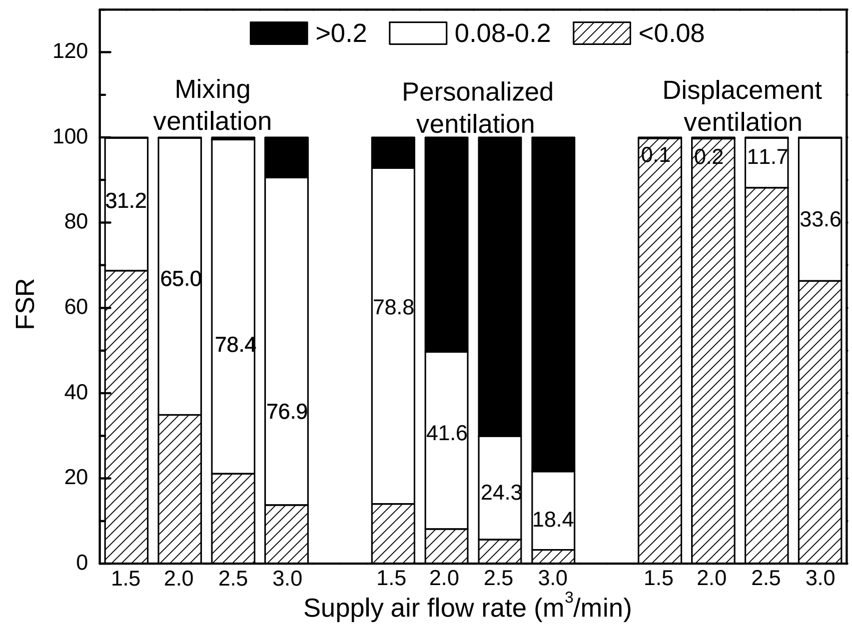
The objective of this investigation was to evaluate the thermal environment and air quality in the sleeping space with the three ventilation systems under different supply airflow rates and air temperatures. Table 3 summarizes the 21 cases that were designed for the study. As shown in the table, the supply air temperature varied from 20 to 26 ^{o}C and the airflow rate from 1.5 to 3 m^3/min. On one hand, a too-low supply air temperature or too-high supply airflow rate would cause a draft. On the other hand, an overly high supply air temperature would be too warm, and an overly low supply airflow rate would cause poor air quality. The cases were designed to identify the best ventilation systems with the most suitable supply airflow rate and air temperature.

Table 3. Air supply temperatures and flow rates for different ventilation systems

Supply air temperature ^{o}C	Supply airflow rate m^3/min	Case number MV	PV	DV
20	2	1.1	2.1	3.1
22	2	1.2	2.2	3.2
24	2	1.3	2.3	3.3
26	2	1.4	2.4	3.4
24	1.5	1.5	2.5	3.5
24	2.5	1.6	2.6	3.6
24	3	1.7	2.7	3.7

Note that the range of supply air temperature was restricted to achieve an acceptable level of thermal comfort. Gong et al. [42] found that the supply air temperatures between 20 and 24 ^{o}C would create an acceptable environment. Considering the potential draft risk caused by close distance between the human body and the supply outlet in the sleeping space, higher temperature is preferred. Therefore this study used a supply air temperature between 20 and 26 ^{o}C. Moreover, the ventilation system should be able to remove the cooling load, this study used a ventilation rate between 1.5 – 3.0 m^3/min which was larger than the need for a sleeping personal ventilation [17].

Figure 8(a) shows that the FSR with low air velocity (≤ 0.08 m/s) was the lowest for PV and highest for DV. This implies that the DV system generated a very low air velocity in the sleeping space. The FSR with high air velocity (> 0.20 m/s) was the lowest for DV and highest for PV. Thus, the air velocity in the sleeping space was very high with the PV system. As the FSR should be as high as possible for air velocity between 0.08 and 0.2 m/s in order to ensure good thermal comfort and CO_2 removal, the MV system seems to be the best. The figure also shows that the FSR for air velocity between 0.08 and 0.2 m/s was a function of air supply flow rate. The MV system performed well at high flow rates, while PV was better at low flow rates. Figure 8(b) shows that the FSR did not change greatly with the supply air temperature.



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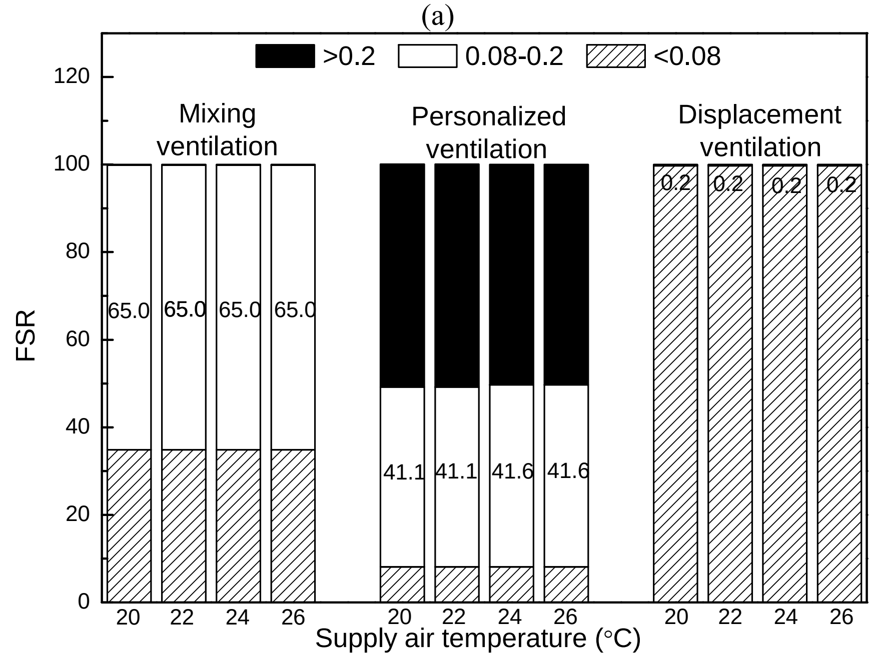
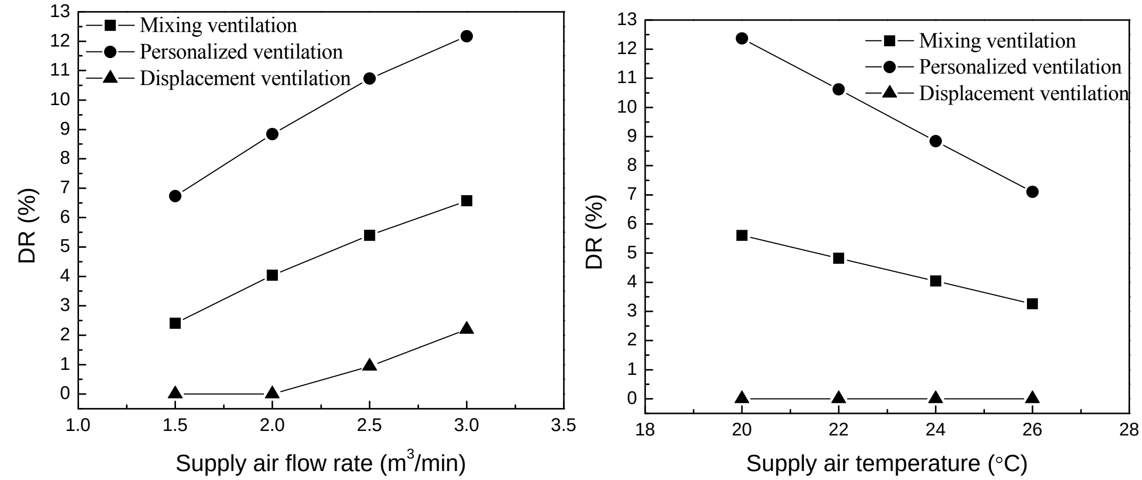


(b)

Figure 8. Comparison of FSR for the three ventilation systems under (a) different airflow rates and (b) different supply air temperatures.

Figure 9 depicts the variation in DR with supply airflow rate and air temperature. The DV system had the lowest DR, and the PV system had the highest. The high DR of the PV system was caused by the short distance between the air supply outlet and the face of the sleeping person. This short distance also made the DR very sensitive to the supply airflow rate and air temperature. Nevertheless, the highest DR calculated for the three ventilation systems was lower than 15%, which was acceptable.



(a) (b)

Figure 9. Comparison of DR for the three ventilation systems under (a) different ventilation rates and (b) different supply air temperatures.

Figure 10 compares the ventilation systems in terms of air quality (MA). The results indicate that the PV system had the youngest MA in the head area, which means that the air was freshest and could easily remove exhaled CO_2. The DV system had slightly poorer performance than that of the PV system, while the MV system had the oldest MA. The results also show that the MA decreased with the increase in supply airflow rate, which was logical. Moreover, this investigation found that the MA did not change greatly with the supply air temperature.

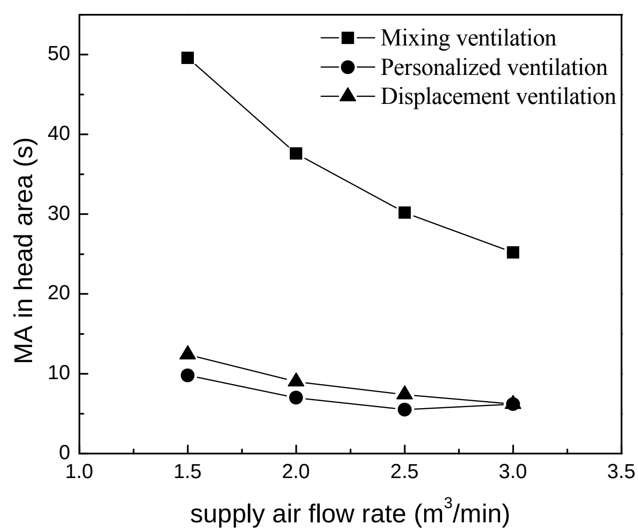


Figure 10. *Comparison of mean age of air in the facial area under different supply airflow rates.*

On the basis of the preceding analysis, this study assigned letter grades to represent the overall performance of the three ventilation systems: A, B, and C. A grade of A represents good performance, B is acceptable, and C is poor. As shown in Table 4, none of the systems received all A ratings according to the evaluation criteria. The PV system seems to be the best, but with a high draft risk. However, Figure 9(b) shows that the DR was acceptable. When the supply air temperature was high, at 26°C, the DR was only 7%, which is very low. The DV and MV systems had similar performance, except that MV had a higher FSR for air velocity between 0.08 and 0.2 m/s, whereas DV had a younger MA.

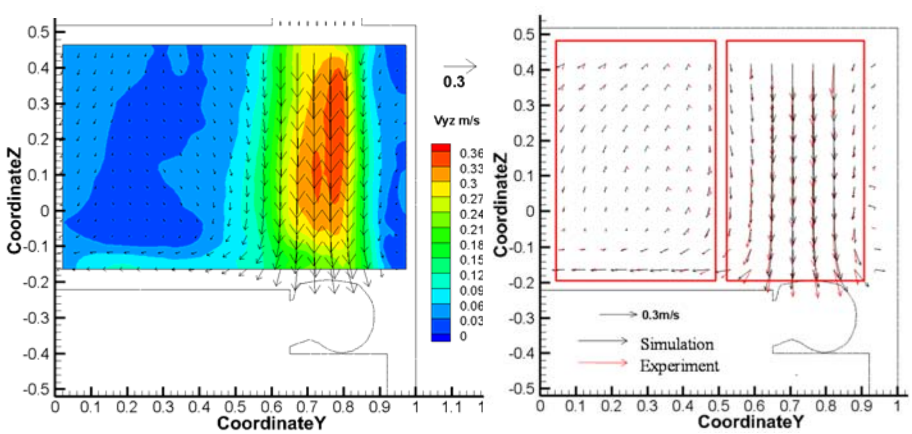
Table 4. Overall performance of the three ventilation systems

Evaluation criteria	Mixing ventilation	Personalized ventilation	Displacement ventilation
FSR between 0.08 and 0.2 m/s	B	A	C
Draft risk	A	C	A
Mean age of air	C	A	B

3.4 Experimental validation of the CFD design

The preceding evaluation process identified the PV system as the best, so it was selected as the design to be validated by experimental measurements in a full-scale test rig. Air was supplied at a uniform velocity of 0.32 ±0.05 m/s, or 1.75 m^3/min (29.2 L/s). The supply air temperature was 20 °C. The SF_6 concentration in the supply air was 0.37 ppm because the concentration in the lab environment was not zero.

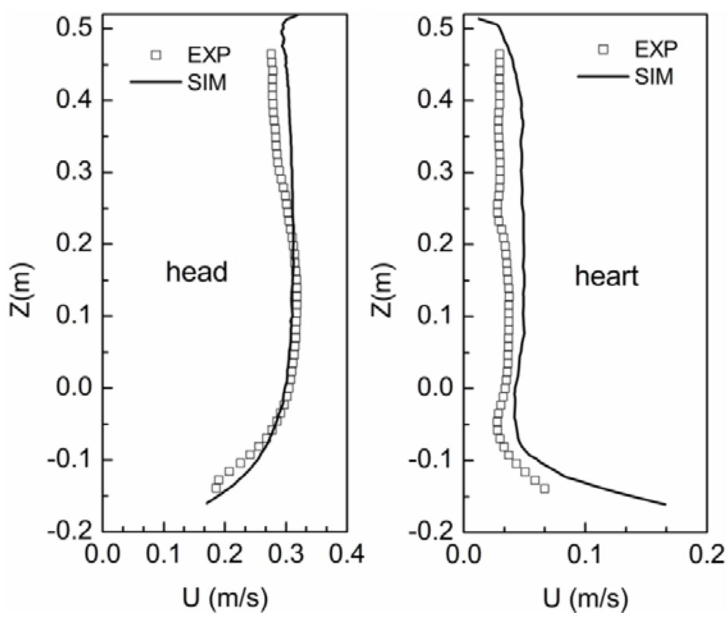
Figure 11(a) depicts the air velocity distribution in the mid-section of the sleeping space as measured by the PIV system. Because the air was blown directly toward the head area, the MA was young. The air velocity near the head area was lower than 0.2 m/s, which would not have created a draft. Figure 11(b) compares the simulated air velocity distribution with the measured data in the mid-section. The airflow on the head region of this section was controlled mainly by the supply air jet, and there is good agreement in this region between the simulated results and measured data. On the heart and feet region of the section, the agreement is not as good because the air velocity in this region was low with strong streamline curvature. The turbulence model did not perform very well for this type of flow.



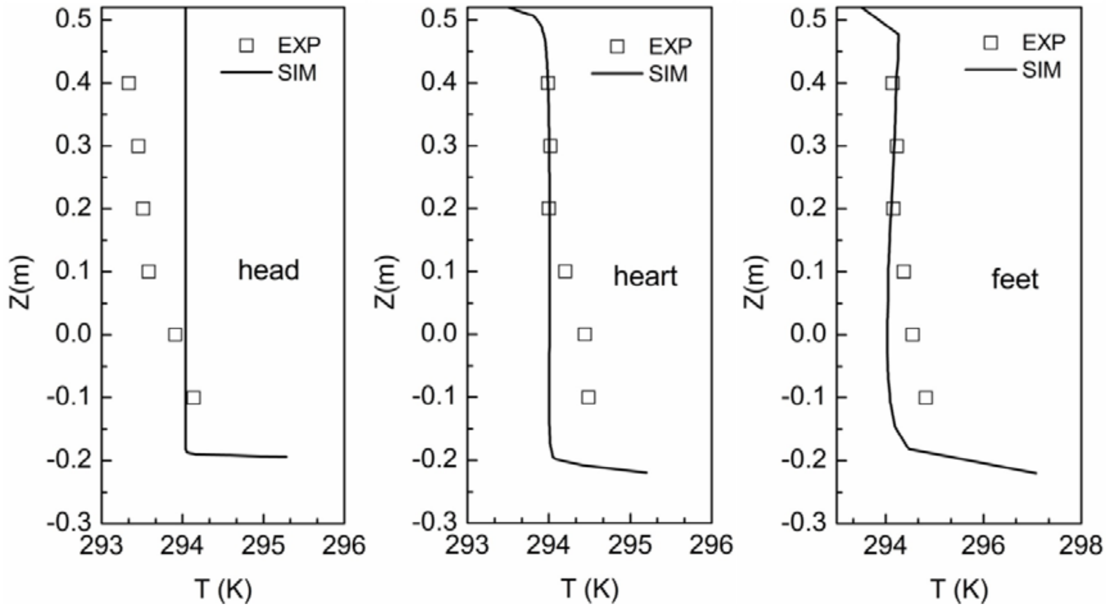
(a) (b)

Figure 11. *(a) Air velocity in the mid-section measured by PIV and (b) comparison between simulated and measured air velocity distributions.*

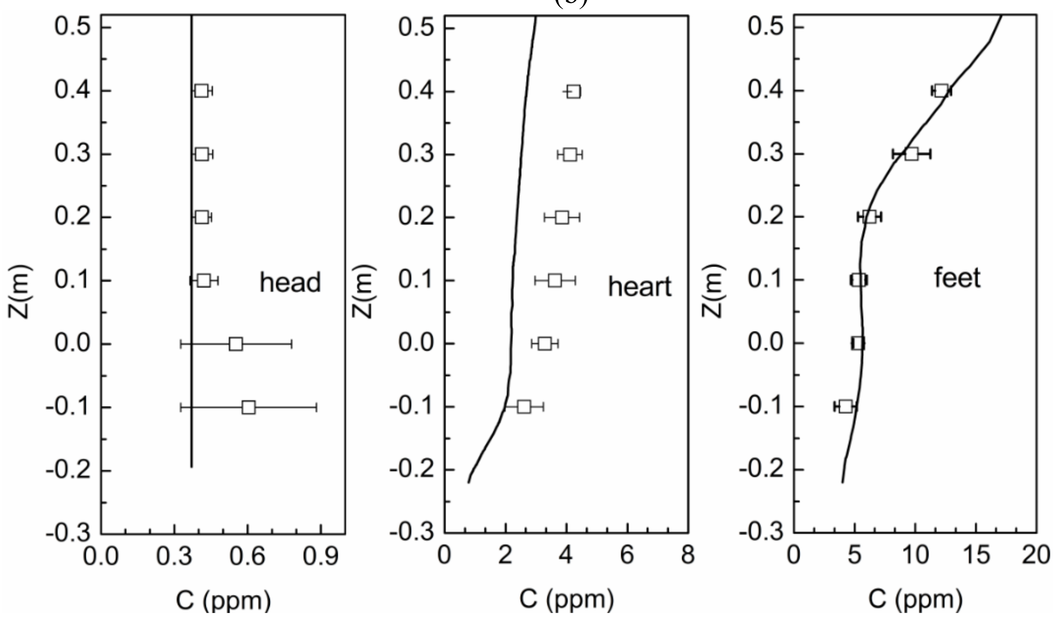
Figure 12 compares the vertical profiles of the simulated air velocity, air temperature, and SF_6 concentration at the head, heart, and foot locations with the measured data. The agreement in air velocity is good at the head position and slightly poorer at the heart location. As discussed above, the turbulence model used could not predict airflow with strong streamline curvature, and thus the agreement in the heart location is not very good. The velocity at the foot location was not available because the laser light sheet did not shine on this area, as shown in Figure 4(a). The temperature was higher near the head and bed than at other locations because of the heat source. Generally, the simulated temperatures agreed well with the measured values. The discrepancies were less than or equal to 0.7 K, which is within the measurement errors. Since the SF_6 concentration measured by INNOVA fluctuated a lot during the experiment, error bars were used in Figure 12(c) to show the fluctuations. The calculated SF_6 concentration profiles and the corresponding measured data have similar trends, and thus the agreement between them is acceptable. However, the differences between the computed results and the measured data are large. Previous studies on room airflow have exhibited a similar problem [34]. The tracer-gas concentration was very sensitive to the sampling position and boundary conditions. A slight difference in measuring location can lead to an error of 50% [34].



(a)



(b)



(c)

Figure 12. Comparison between simulated vertical profile and measured data at head, heart, and foot locations: (a) air velocity, (b) air temperature, and (c) SF_6 concentration.

By comparing the calculated and measured results, the relative average error for the air velocity was 18.2%, for air temperature 0.1% and for SF_6 concentration 28%. The comparisons shown in Figures 11 and 12 confirm that the CFD results are reasonably accurate. Thus, the optimal design obtained by CFD is reliable and can be used for the future design of sleeping spaces.

4. Discussion

This study found the PV system to be the best among the three systems studied. This conclusion is similar to previous findings for office buildings [43] and bedrooms [14-16]. In order to reduce the energy requirement for conditioning a sleeping space, it is essential to explore the minimum ventilation rate at which air quality is still acceptable. To provide satisfactory air quality, the ventilation system should effectively remove CO_2 from the sleeping space. The CO_2 concentration can be calculated from the following equation under perfect mixing conditions:

$$V\frac{\mathrm{d}C}{\mathrm{d}\tau}=QC_S+\dot{\mathrm{m}}\text{-}QC \tag{4}$$

where V is the volume of the sleeping space, C the CO_2 concentration at time τ, Q the ventilation flow rate, C_s the CO_2 concentration of the supply air, and $\dot{m}$ the generation rate of CO_2 in the sleeping space. The initial CO_2 concentration is C_1 at time $\tau = 0$, and thus the concentration C_2 at time τ = τ can be calculated from

$$C_2=C_1\exp\left(-\frac{Q}{V}\tau\right)+\left(\frac{\dot{\mathrm{m}}}{Q}+C_S\right)\left[1-\exp\left(-\frac{Q}{V}\tau\right)\right] \tag{5}$$

When $\tau \to \infty$, $C_2 = \frac{\dot{\mathrm{m}}}{Q}+C_S$. For a sleeping space with the assumption of C_1 = 500ppm and $\dot{m}$ = 13L/h [43], the supply air flowrate should be at least 7.2 L/s in order to maintain C_2 at less than 1000 ppm for a long duration. Since the volume of the sleeping space was 1.68 m^3, the minimum ventilation rate was 15.5 h^{-1}, which is much higher than that in buildings because of the small volume of the sleeping space. Although this air exchange rate sounds very high, the ventilation rate per person is comparable between the sleeping space and building.

Figure 13 shows the CO_2 concentration in the inhalation zone (the cube shown in the figure) at various supply airflow rates for the PV system. To maintain a CO_2 concentration under 1000 ppm in the inhalation zone, the supply airflow rate should be at least 25 L/s. This rate is necessary because it is difficult to remove the zero-momentum CO_2 source. The non-uniform CO_2 distribution required a much larger supply airflow rate than one normally found in a room. Thus, the assumption of uniform CO_2 for the PV system was not valid. The curve in Figure 13 is not smooth because the airflow in the sleeping space was non-linear.

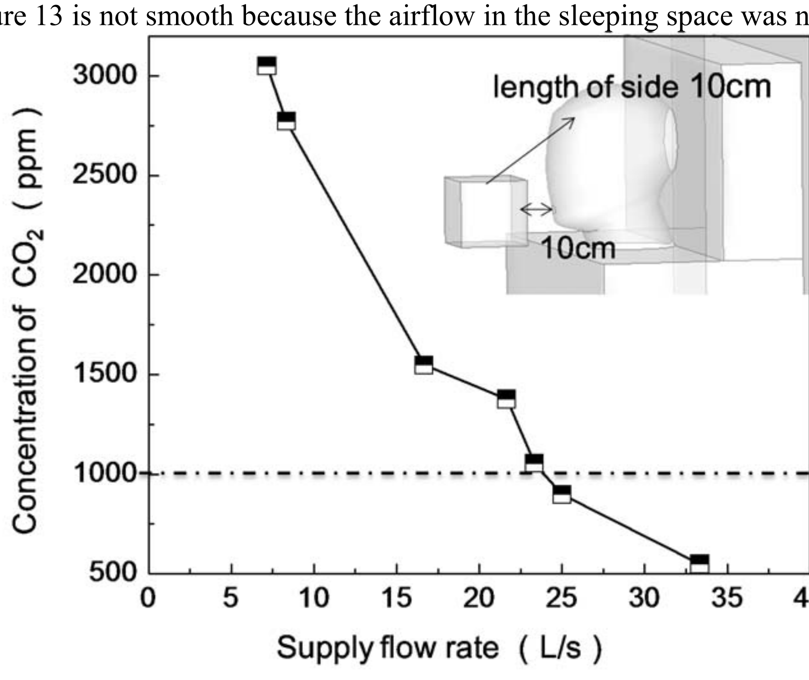


Figure 13. *CO_2 concentration in the inhalation zone under various supply airflow rates for the PV system.*

It should be noted that this study investigated only one sleeping posture. Different postures could lead to different results and should be studied further in the future.

5. Conclusions

This study used a verified CFD method to evaluate the performance of three ventilation systems in a small sleeping space on the basis of facial-area speed ratio, draft risk, and mean age of air. The investigation led to the following conclusions:

The CFD method is an excellent tool for evaluating the air distribution in the sleeping space. The CFD results enable us to evaluate air quality quantitatively in terms of the mean age of air, and thermal comfort in terms of draft risk and facial-area speed ratio.

The personalized system had the highest facial-area speed ratio for air velocity between 0.08 m/s and 0.20 m/s and provided the freshest air, in comparison with the mixing and displacement ventilation systems. Thus, the personalized ventilation system can provide the best air quality with acceptable draft risk.

The personalized ventilation system identified by CFD was further validated by measuring the air velocity distribution in the mid-section of the sleeping space with particle imaging velocimetry, and by measuring the air temperature with thermocouples and the SF_6 concentration with a photoacoustic gas analyzer in the head, heart, and food locations. The agreement between the computed results and the measured data was generally good, with some discrepancies in the recirculation zone, where the turbulence model did not perform well.

To maintain the CO_2 concentration at less than 1000 ppm in the inhalation zone, the ventilation rate should be at least 25 L/s. The corresponding air exchange rate was 15.5 h^{-1}. Although this rate sounds very high, the ventilation rate per person was comparable to that in buildings.

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